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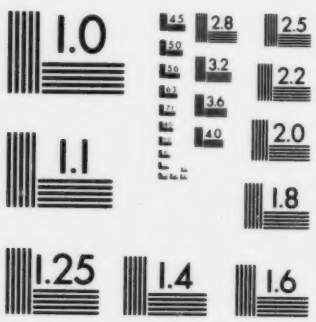

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THE DESIGN, CONSTRUCTION AND COMMISSIONING OF A WIDE PLATE TEST FACILITY

Welding Institute of Canada Report RC203/1/86

FINAL REPORT

Prepared by:
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Work on this project was conducted under the auspices of the Canada Centre for Mineral and Energy Technology, Energy, Mines and Resources Canada.

DSS File Number: 23SQ.23440-4-9078
DSS Contract Serial Number: OSQ84-00058
CANMET Scientific Authority: Dr. B. Faucher

Approved by:

Dr. A. G. Glover,
Director of Technology

October 1986

MICROMEDIA

ABSTRACT

The need to assess the risk of fracture at low temperatures is essential in view of forthcoming requirements for Canadian frontier oil and gas developments. As part of the overall Canadian effort to study these requirements the Welding Institute of Canada and the University of Waterloo have jointly designed, constructed, and commissioned a Wide Plate Test Facility to study the fracture characteristics of welded components.

The facility has been designed and built and shown to be fully operational. The test rig is a rigid machine intended for studying crack initiation, and short crack propagation and arrest. The rig can test plates up to 1220 mm in width with a maximum load capability of 4000 T. The test plates can be locally cooled and the facility is fully instrumented. A commissioning fracture test has been successfully completed.

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Suite aux besoins créés par le développement de l'industrie canadienne du gaz et du pétrole, il est essentiel de pouvoir évaluer quels sont les risques de rupture à basse température.

Afin de participer à l'effort canadien entrepris pour définir quels sont ces besoins, l'ISC et l'Universite de Waterloo ont conçu et construit en commun un <<banc d'essai sur grande plaque (Wide plate test facility)>> pour étudier les caractéristiques de ruptures des assemblages soudés.

Ce banc d'essai est maintenant tout à fait opérationnel. Il consiste en un assemblage rigide prévu pour l'étude de l'amorçage de fissures, la propagation de fissures courtes et leur arrêt.

Le banc peut mettre à l'épreuve une plaque atteignant 1220 mm de large, avec une capacité maximale de charge de 4000 tonnes. Les plaques d'essai peuvent être refroidies localement et le banc est complètement instrumenté.

Un contrat d'essai de rupture a déjà été complété avec succès.

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1.0 Introduction

The concern for adequate material and welding technology associated with the design, fabrication, inspection, and operation of structural components of offshore platforms, caissons and Arctic vessels has obviated Supply and Services, Canada to develop the capability of addressing these needs. As part of the overall Canadian effort to study these problems, the Welding Institute of Canada and the University of Waterloo have jointly designed, constructed, and commissioned a wide plate test facility to study the fracture characteristics of welded components. This report describes the design details and fabrication of the wide plate machine and concludes with the successful testing of the wide plate test facility.

2.0 Canadian Requirements

The need to assess the risk of fracture at low temperatures is essential in view of forthcoming requirements for Canadian frontier oil and gas development. In these regions the severe environmental conditions will necessitate thick section weldments in a range of structures and support vessels. For such components, it will be necessary to review those aspects which determine the fracture risk, i.e. material properties and toughness, stress levels and defect sizes, and the relation of these to codes and standards. The structures and vessels to which the wide plate test results will apply are subjected to both static, intermediate and dynamic loading. Potentially a large number of service fractures could result from static loading and hence the slow strain rate capabilities of the proposed facility will permit investigation of this failure mode.

Traditionally, workmanship standards relating to the defect acceptance levels in welds used in fabrication, are not based on the engineering significance of defects, but rather are historical and simply reflect what is reasonably achievable with the welding process used. Tests such as the Charpy V-notch and drop weight tests have been used for many years as a measure of toughness acceptance criteria. However these do not adequately include actual stress distributions around the defect and the effect of local brittle zones in weldments. Although the acceptance levels derived from these tests provide a given standard of quality, they do not provide any quantitative information on the actual failure risk. In the event that the workmanship standards are not achieved or where there is damage during service or the growth of weld defects due to corrosion or fatigue, decisions to repair, or not, must be based on an engineering assessment of the significance of the damage or weld defect. The study of fracture mechanics has allowed a methodology for this assessment to be developed in the United Kingdom which has been formally documented. The fracture mechanics tests used in the assessment have been indexed to full scale tests on welded plates and the design curve used is considered to be conservative [8].

A similar approach has been taken in Canada for the evaluation of defects in pipeline girth welds. CSA Standards have recently been amended to allow the use of an engineering critical assessment for defects that exceed workmanship standards, and to consider it as a design approach. As part of this work the Welding Institute of Canada undertook full scale tests on large diameter pipes in which girth welds containing known defects were subjected to stress by bending of a 12.2 m length of pipe. The results were compared with small laboratory fracture mechanics tests on various regions of the weld to establish the degree of conservatism in the existing proposed design curves, to determine how residual stress might be included and to determine the effects of the shape of the defect. In some cases it was found that the design curve was overly conservative and it is believed that modification of the proposed methodology is warranted, [9, 10, and 11].

To date, no full scale tests on welded plates of the type considered for marine vessel application in the Arctic have been carried out in Canada. It

is considered that such tests are essential to provide a proper data base for the development of fabrication standards, weldment material toughness requirements and of particular importance is the defect tolerance in welded plates in the thickness range 50-75 mm. In marine vessel construction such welds would not be stress relieved whereas in pressure vessel applications plates of such thickness would normally be stress relieved. It is known that thermal stress relief has a significant effect on the risk of fracture both by changes in weldment properties and by lowering residual stresses [12]. In many other critical structures, (e.g. North Sea structures) plates thicker than about 40 mm are often required to be thermally stress relieved. An Arctic vessel, therefore, with thicker section welded plates in the as welded condition, represents a unique situation.

One approach to the study of fracture behaviour of heavy section plate weldments for offshore applications is to carry out wide plate tests. While development and experience elsewhere in the world, ie. the North Sea, have provided a good basis for design and construction the particular features of the Canadian offshore, lower temperatures, ice loads, icebergs and storm frequencies need special attention. The results of wide plate testing of Arctic grade steels will be compared to small scale fracture mechanics tests (such as CTOD tests) as in previous work with a view to obtaining empirical relationships [13]. These test results will be invaluable in the ongoing development of fracture control plans for the Canadian offshore platforms, icebreaking LNG and crude carriers. The establishment of a wide plate testing facility in Canada is therefore a key requirement in achieving high reliability welded fabrications for the most demanding service conditions [14 - 17].

3.0 Background of Wide Plate Tests

The concept of wide plate tests has been in existence for some 30 years and evolved principally as a result of ship fracture problems in World War II. However, due to the expense and time required to conduct such studies the number of investigations pursued was relatively small and hence limited data is available. Small scale fracture tests have enabled a more convenient study

of fracture initiation, propagation, and arrest, but they are limited in their ability to study the effects of residual strain fields. Previous studies of brittle fracture propagation in wide plates tests using pre-strained plates 610 mm wide indicated that an induced residual strain field had a marked effect on the propagation characteristics of a fracture, [1] and are therefore necessary to confirm full scale behaviour. However, the magnitude and the extent of the strain field was necessarily limited by the size of specimen and it was not always possible to easily separate the effects of the properties, size and residual strain on fracture. Hence further studies have used plate widths ranging from 914 mm to 1830 mm to facilitate a study of the factors influencing fracture propagation through a residual strain field and to quantify the effects of the material behaviour and residual strain. The primary purpose of such investigations were to study the propagation characteristics of a brittle fracture by examining the fracture speed and the strain field associated with the tip of the propagating crack [2].

Various test piece designs have been utilized in wide plate tests, notably the David Taylor Model Basic (DTMB), Standard Oil Development Company (SOD), Robertson, Wells and Japanese double tension test in order to investigate different fracture phenomena and concepts such as initiation, propagation and arrest [3]. To date there are approximately twelve wide plate facilities throughout the world with capacities ranging from 500 - 10,000 metric tons (see Table 1 for details). The lower capacity machines are normally more flexible with vertical columns, notably at the Universities of Illinois, California and Lehigh, whilst the higher capacity machines are more rigid machines, specifically in the UK, Europe and Japan [3].

Results from the Wells wide plate tests [4, 5, 6], which were established on a rigid type of machine, were used to establish toughness criteria for pressure vessels and were, and still are, incorporated in British Standard BS5300. Data from these previous tests is only available for basic C-Mn steels and not the grain refined micro-alloyed steels now available for structural marine applications. These results were utilized primarily for an initiation approach to fracture.

With regard to the concept of crack arrest, a simple fracture mechanics approach has been successfully applied to the analysis of the experimental results of several conditions of brittle crack arrest using medium-sized wide plate specimens [7]. This analysis is based on the arrest toughness concept and the consideration that the dynamic aspect of the stress intensity factor or strain energy release rate can be taken into account. In this case, the crack length is relatively small, and can be analyzed as equivalent to an arresting crack or a crack running at a speed near to the lower critical velocity of a brittle crack [7]. Previous work on wide plate tests 900 mm wide has indicated that crack arrests were only produced under isothermal conditions when the tensile residual stress field arising from the weld is allowed to remain, that is, the specimens are not given a mechanical or thermal stress-relief treatment. The arresting conditions are apparently produced by a downward gradient of crack extension force along the crack path from the weld at the center of the plate. Wells [4] found that crack arrest occurred only with spontaneous initiation (cooling without external loads) or with external loads giving rise to average stresses less than about one third of the yield. While it is possible to investigate brittle fracture arrest characteristics for short arrested cracks with plates of this size, it has been shown [3] that in order to study the arrest of long cracks, plates of at least 2500 mm wide would have to be considered, thus requiring a test rig with at least a 9000 metric ton capacity. This was considered beyond the scope of the present program, hence the design was limited to study initiation and short crack arrest.

4.0 Design Requirements

The design of the Wide Plate Test Facility was carried out by Professor R. J. Pick and A. DeGroot at the University of Waterloo. The concept of the facility was to be able to test a range of plate widths and thicknesses on material with high yield strengths. The initial design concept is shown in Figure 1 and is based on loading the test plate through attached end lugs. Preliminary design calculations (Appendix A) reviewed the stresses in the lugs, the pistons (capsules), end frame, and distribution of load in the plate. This latter calculation was performed on the basis of a 1820 mm (72

in.) wide plate, which showed a 10% variation in load across the width of the plate. As a result of this variation a plate width of 1220 mm was considered and it was shown that the variation was reduced to less than 5%. Since it has also been shown [3] that a residual strain field representative of full-sized structures can be achieved in plates of 1220 mm width, this size was chosen for the final design. The end frames were designed to have sufficient stiffness to ensure even distribution of load. A further consideration was given to the maximum plate length which can be used. A decision was made to design on a maximum length of 1980 mm and a minimum length of 1320 mm, based on the capsule size and potential cooling requirements.

These preliminary decisions resulted in a rigid machine being designed with a capacity of 4000 T that would allow the study of crack initiation, and short crack propagation and arrest to be made. The final detailed design drawings are given in Appendix B and consist of:

- D1212-1 A Assembly 1000 Ton Capacity Jack
- D1212-2 B Details of 1000 Ton Jacks
- D1212-3 A Details of 1000 Ton Jacks
- D1212-4 C Jack Support Details
- D1212-5 A Base Frame and Portal
- D1212-6 B General Assembly
- D1212-7 B Plate Specimen
- D1212-8 B Shock Absorber Assembly and Details
- D1212-9 A Welding Jig

The capability of the facility in terms of failure stress versus capsule capacity is shown in Figure 2, which indicates various thickness limitations for maximum failure stress at the 4000 T load.

In order to limit the size of the cross beams it was decided that the piston of each capsule should only be about 250 mm in diameter. For the development of loads of 1000 T each, this meant an internal pressure of 207 MPa would be required. No standard capsules are available for these loads and a special design was utilized. Initial designs of a shrink fit were changed to that of a high strength forging and the design given in D1212-1. This design also

permitted a stroke of at least 50 mm to be achieved. This stroke is required because of the high ductility at failure of present and proposed materials, and can be increased by modifications to the capsules.

The general assembly of the facility, D1212-6, shows how it was possible to construct a compact facility through the use of high hydraulic pressure capsules. The assembly is mounted on a base frame which provides a restraint system for retaining the fractured plate, and on which are also attached instrumentation and cooling equipment. A sub frame on either side of the test plate supports the hydraulic capsules and allows for their easy removal to access the test specimen. The piston length can be altered to suit the test plate length by adding or removing various shims. The hydraulic system is designed so that the four capsules will operate together but are each individually driven by a pneumatically controlled Rousseau pump. This balanced arrangement, together with the test plate being freely supported eliminates any bending effects, provided the test plate is carefully aligned.

The end frames are constructed from structural steel supplied to CSA G40.21 and fabricated to CSA W59 using electroslag welding. The final design utilized a contoured solid end plate with four machined lugs welded onto it. The restraint system, which consists of a series of crushable pipes, was located at the end of one of the frames.

5.0 Fabrication

After finalizing the design and drawings, ordering of the components commenced and decisions on the fabrication route were made. The base plate and support material were supplied by Algoma Steel and the remaining components purchased. The fabrication of the capsules and support systems was carried out by the University of Waterloo machine shop, while machining of the end plates was conducted at MetriCan Manufacturing, Oakville. The detailed fixturing of the assemblies to the base plate, welding and assembly took place at the Welding Institute of Canada.

Welding of the lugs to the main end plates was the most critical aspect during

assembly of the test rig. These eight welds were made using consumable guide electroslag welding. Two guides were used with oscillation of the head at 600 Amps, 47 V with a welding speed of approximately 5 mm/min. A complete ultrasonic inspection after welding confirmed that the welds were acceptable, with the exception of some lack of fusion at the edges. These regions were gouged out and the weld/lug assembly profiled using E7018 electrodes. Following successful completion of welding the two assemblies were completely normalized. The lugs were then machined square to the piston assembly and location points for the pistons machined into them.

Using a trial plate the facility was assembled so that the hydraulic and instrumentation connections could be made. An overview of the facility is shown in Figure 3, details of the hydraulic connections areas in Figure 4 and the collapsible tube restraint system is shown at the end of the test piece in Figure 5.

6.0 Cooling System

Several ideas were considered for the cooling system and two were evaluated.The first relied on using blocks of dry ice placed in close contact to the plate over a width of 30 cm and depth of 30 mm, and the whole plate was surrounded by an insulating box. This trial was carried out on a 75 mm thick plate and temperatures between -25° and -30°C could be achieved. The second technique utilized a circulating system of liquid nitrogen around the test plate. Again the trials were conducted on a 75 mm thick plate (Figure 6) and temperatures down to -60°C could be readily achieved, with temperature variations within 5°C. Better control can be achieved with improved flow patterns using the nitrogen vapour.

7.0 Commissioning

In order to validate the test frame several commissioning trials were carried out. The first series looked at the functioning of the capsules at low and intermediate pressures. The second looked at the behaviour of the frame with a trial plate in it and the third took the plate to failure to assess the overall behaviour of the test rig and restraint system.

Initial trials were carried out on the capsules themselves following fabrication at pressures of up to 4.1 MPa (600 psi) to check for is leakage at the seals. No leakages occurred and the capsules were installed into the test frame.

The next series of tests concentrated on evaluating the loading of the frame and distribution of load within a test plate. A 25 mm thick test plate was SMA welded into the end lugs using a balanced procedure and referenced locations to check alignment. This technique produced some distortion in the test plate but the end lugs remained in-plane. The test plate was instrumented with strain gauges as shown in Figure 7, with matching gauges on either side of the plate. The fillet weld regions of the end lugs were also strain gauged, together with the 12, 3, 6 and 9 o'clock regions of the cylinders. These series of gauges were to assess whether there was any bending at the lugs or any undue bending of the cylinders.

Tensile tests on the test plate gave measured strengths of 340.69 MPa (49,400 psi) yield and 537.9 MPa (78,000 psi) ultimate. For the test plate dimensions of 940 mm by 25.4 mm, the theoretical elastic stress and strain values versus jack pressure are given in Figure 8.

Initial loading of the plate did not give a very good correlation between theoretical and actual strain. The reason for this was that the capsules were being asked to respond too rapidly. The capsules initially only move extremely slowly and need to be brought into tight contact with the end lugs prior to the test loading cycle. A jack pressure of 0.35 to 0.5 MPa (50 to 75 psi) is sufficient to overcome the internal friction of the cylinders. The results of the first and subsequent tests are shown in Figure 9. These show that the variation from the theoretical curve was reduced from 12% to 7% by initial slow loading. The variation of 7% is caused by the deformation in the plate from the welding operation. Subsequent loading further straightened the plate and reduced the variation to less than 5%. Strain around the lugs showed some variation with the maximum of approximately one-quarter of the yield strain. The strain in the pistons showed variation around the circumference, indicating some out-of-plane loading. The response of the plate (top to bottom) on loading is shown in Figure 10 with a comparison of the surface strains on

both sides. The results include both manual strain measurements and computer results from the data acquisition system. There is some variation between the manual and computer results because of the program used to generate strains and this has since been corrected. The variation from the top to the bottom of the plate is acceptable, however the variation from one side to the other was not. This latter variation arises from the bending introduced in the plate as a result of the welding operation. Being satisfied that this would not adversely affect the objectives of the commissioning test, loading was continued as planned.

These initial commissioning tests proved the operation of the capsules and the test frame and validated the control system. A minor modification to the hydraulic controls was made to enable better control to be achieved during the loading cycle.

The amount a frame defects under load is usually expressed in terms of the deflection at full load or spring rate. The load frame stiffness (lb/ins, or kg/mm) for the wide plate test was based on the analysis of the four rams under compressive loading during the commissioning tests. The analysis had already checked that the bars would not be subjected to buckling and hence the spring constant of the axially loaded bars is

$$K = \frac{AE}{\ell}$$

The deflection of the pistons was monitored during loading and the deflection in the plate also noted. These deflections were also checked at a later stage at higher loadings on a thicker plate. Based on these results the spring constant at full load was 80×10^6 (1.44×10^6 kg/mm).

A failure test was proposed as the final demonstration of the capability of the test rig. It was proposed to use the same test plate as before with a through-thickness notch located in the centre of the plate. It was proposed to carry out the test at -25°C, requiring small scale CTOD tests to be carried out at this temperature. All the tests gave maximum load behaviour with a CTOD value of 0.32 mm (0.0126 in.) at -25°C. The 250 mm (9.84 in.) long defect was inserted in the centre of the plate using a hole extended at either end by a saw cut and finally by slitting to give a 0.12 mm (0.005 in.) notch. The predicted failure strain was determined using the technique of PD6493.

where $\varepsilon_t = \frac{\delta}{2\pi\bar{a}} + 0.25\,\varepsilon y$

where ε_t = tolerable failure strain

ε_y = yield strain

δ = CTOD toughness

$\bar{a}$ = defect parameter

For the present test.

ε_y = 0.001647 strain

δ = 0.32 mm

$\bar{a}$ = 125 mm ($\ell/2$)

hence ε_t = 0.00082 strain

The plate was initially cooled by using dry ice but later liquid nitrogen was used to maintain the plate at temperature. The average temperature of the test region at the start of the test of -29°C decreased negligibly to -30°C at the completion. The test took 40 minutes to complete with failure occurring at a jack pressure of 36.2 MPa (5250 psi), corresponding to a plate strain of 1650 μstrain. Based on the above calculation the safety factor on strain (failure strain divided by tolerable strain) was 2.01 which is in agreement with the basis of the PD6493 COD design curve and with previous full scale pipe fracture tests.

Problems arose with the attachment of the clip gauges. Previously this had been successfully done using adhesive bonding of the knife edges and used on tests at temperatures of -45°C. Early on in the loading of the plate one knife edge became detached and hence future tests will use modified knife edges with a screw attachment.

The restraint system also proved successful and at these relatively low energy release rates no permanent deformation of the rings occurred. A photograph of the end result of a higher energy release rate test is shown in Figure 11. Higher energies can be absorbed by using more smaller diameter rings and/or thicker sections.

This initial test has emphasized the importance of alignment of the test plate and hence a decision was made to fabricate an alignment rig (drawing D1212-9A). A current program using the welding alignment rig has shown that its use has eliminated the problem of distortion in the testpiece.

8.0 Conclusions

The objective of this research program was to design, construct and commission a wide plate test facility. The rig has been designed and built and has been shown to be fully operational. This has allowed a national facility to be developed that will be available to Canadian industry and government, owned and operated by the Welding Institute of Canada. The facility will be used to study the fracture characteristics of welded components.

The design has been carried out by the Mechanical Engineering department of the University of Waterloo. The test rig is a rigid machine intended for studying crack initiation and short crack propagation and arrest. The rig can test plates up to 1220 mm in width, 1320 to 1980 mm in length with a maximum load capability of 4000 T. The load is applied through four 1000 T capsules, each having a 50 mm stroke. The stroke can be changed by modification to the capsules. The hydraulic capsules are in a balanced arrangement and are controlled through a pneumatically driven pump and pressure transducers. The test plate is freely supported to eliminate bending. Preliminary tests have shown that the test piece needs to be carefully aligned and this has necessitated the fabrication of an alignment rig. Various cooling systems have been tried and the most successful was one that utilized recirculating liquid nitrogen vapour through a controlled cooling box. The restraint system has been validated and its capacity can be increased by varying the crushable pipe diameter and thickness.

Loading of the test plate was shown to behave as predicted elastically once frictional effects had been overcome. Changes will be made to the control system to improve the low pressure control.

9.0 Acknowledgements

This program was funded jointly by the Welding Institute of Canada and Energy Mines and Resources. The authors would like to thank Ari De Groot who supplied much of the engineering design. Also the time and effort of Dirk Varo, Neil Sweetman, Paul Koornneef and Azael Aguire, without whom the electroslag welding may never have been done. Acknowledgements are also given to other members of the University and Institute staff who contributed to the program, together with Bernard Faucher who monitored the program.

10.0 References

1. Rolfe, S.T. and Hall, W. J., "Brittle Fracture Tests of Two Feet Wide Steel Plates Containing Residual Compressive Strain", Welding Research Supplement, Welding Journal, April 1959.

2. Barron, R. W. and Hall, W.J., "Brittle Fracture Tests of Six Foot Wide Prestressed Steel Plates", Welding Research Supplement, Welding Journal, Sept. 1960.

3. Kihara, H., Hall, W.J., Soete, W. and Wells, A.A., "Brittle Fracture of Welded Plate", Prentice-Hall publication, 1967.

4. Wells, A.A., "Brittle Fracture Strength of Welded Steel Plates", British Welding Journal, May 1961.

5. Wells, A.A., "Testing Thick Plates for Brittle Fracture", Nuclear Engineering 69:7, 1962.

6. Rosenstein, A.M. and Labahn, J.D., "Brittle Fracture of Welded Steel Plates", Welding Research Supplement, Welding Journal, Nov. 1967.

7. Hahn, G.T. and Kanninen, M.F., ASTM Special Technical Publication 627, "Fast Fracture and Crack Arrest", 1977.

8. British Standard Publication PD6493-1980, Guidance on Some Methods for Derivation of Acceptance Levels for Defects in Fusion Welded Joints.

9. Welding Institute of Canada Report: RC75/2/83, May 1983.

10. Glover, A.G., Coote, R.I. and Pick, R.J., "Engineering Critical Assessment of Pipeline Girthwelds", Fitness for Purpose Validation of Welded Construction, Welding Institute UK, London, 1981.

11. Glover, A.G. and Coote, R.I., "Full Scale Fracture Tests of Pipeline Girth Welds", Conference ASME, Circumferential Cracks in Pressure Vessels and Piping, Portland, June 1983.

12. Soete, W., "The Wide Plate Test", Metal Construction, February 1980.

13. Marique, C. and Bragard, A., "Weldability of Structural Steels - Assessment by Wide Plate and COD Tests", Metal Construction, June 1979.

14. Pellini, W. S., "Principles of Fracture Safe Design", Part 1, Welding Research Supplement, Welding Journal, March 1971.

15. Wells, A.A., "A 600 Ton Test Rig for Brittle Fracture Research", British Welding Journal, Jan. 1956.

16. Irwin, G.R. and Kies, J.A., "Fracturing and Fracture Dynamics", Welding Journal 1952, Vol. 41.

17. Greenspan, M., Journal Research National Bureau of Standards, 1943 Vol. 31.

TABLE 1

Details of major wide plate facilities world wide.

Location	Maximum Capacity Metric Ton	Type of Rig
University of Illinois	1360 tons	Flexible (Vertical)
University of California	1360 tons	" "
Lehigh University	500 tons	" "
Europe		
Queens University, Belfast	4,000 tons	Rigid (Horizontal)
UK Welding Institute	4,000 tons	" "
Centre de Recherches Metallurgiques, Liege	400 tons	Rigid (Vertical)
University of Gent Belguim	6,000 tons	Rigid (Horizontal)
Czechoslovakia	8,000 - 10,000 tons	Not available
Vickers-Armstrong Ltd. UK	4,000 tons	Flexible (vertical)
Japan		
Ship Research Institute	4,000 tons	Rigid (Horizontal)
Nippon Steel	Information not available.	

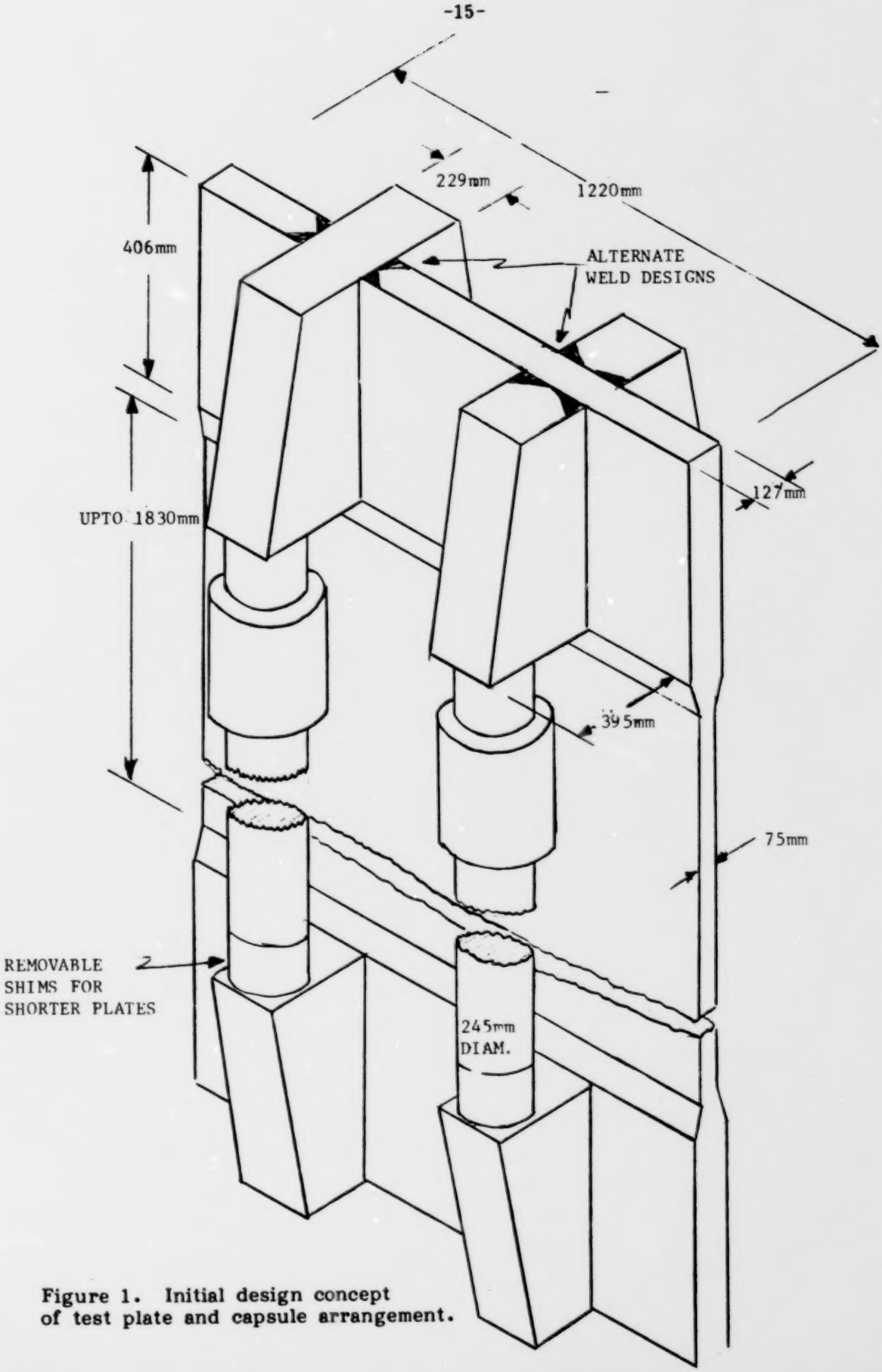


Figure 1. Initial design concept of test plate and capsule arrangement.

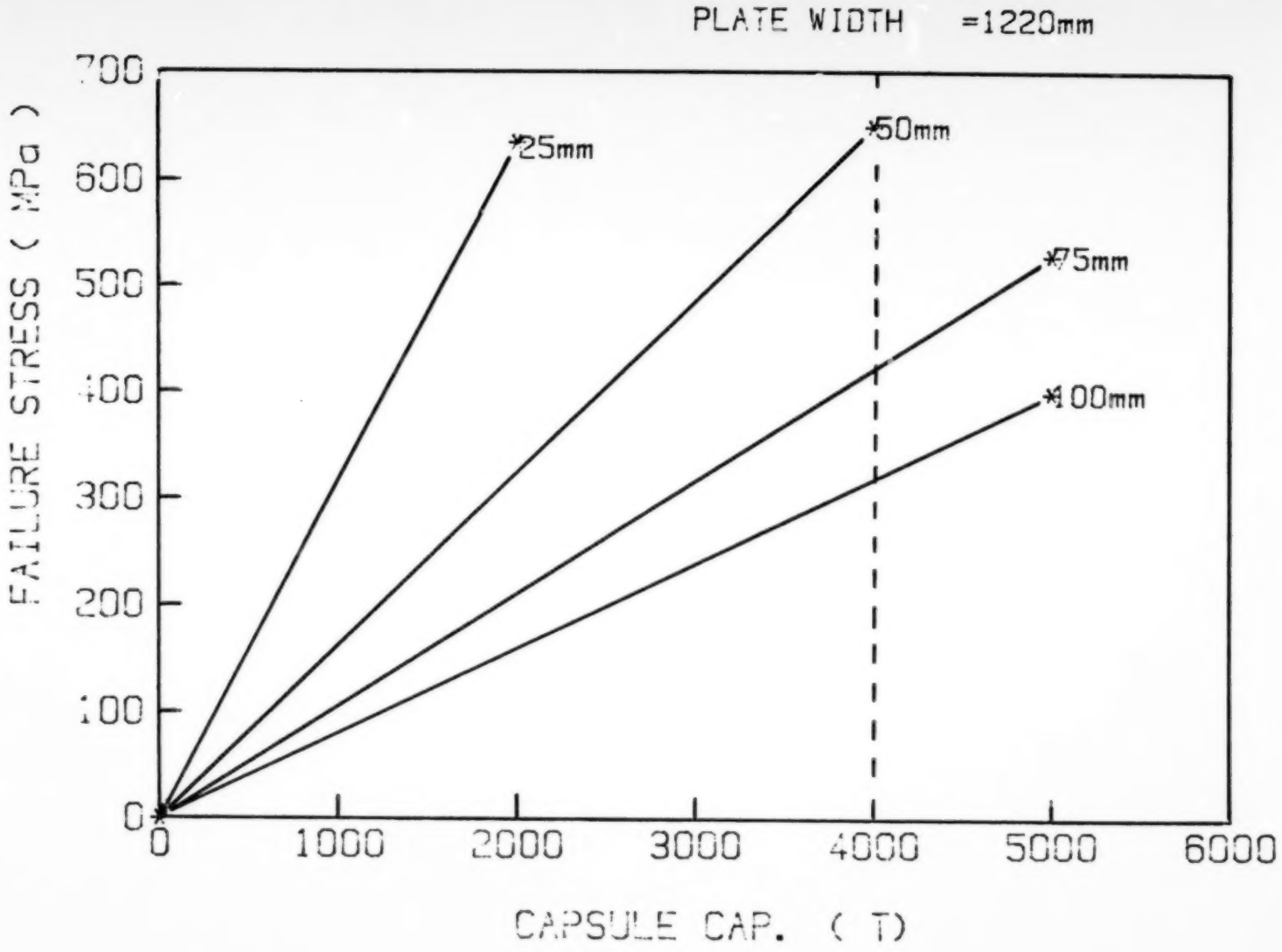


Figure 2. Effective capacity of the machine, in terms of failure stress and capsule load for different plate thicknesses.

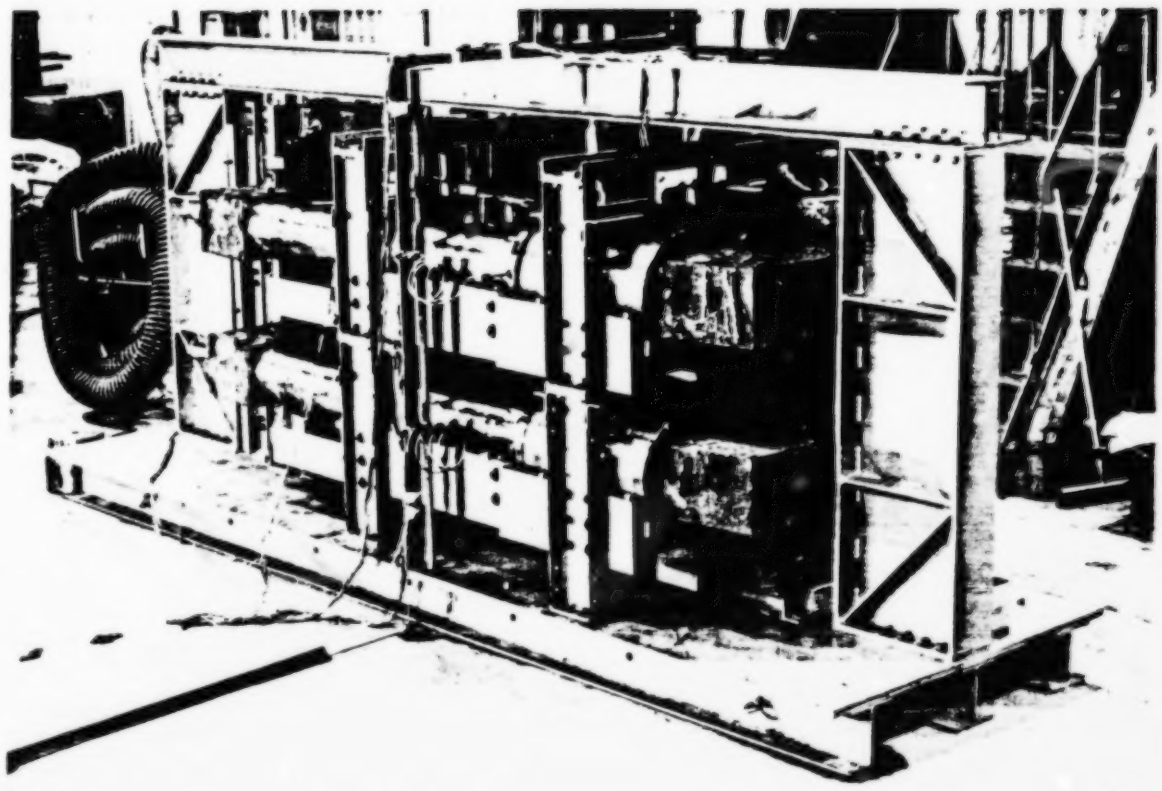


Figure 3. Overview of assembled test facility.

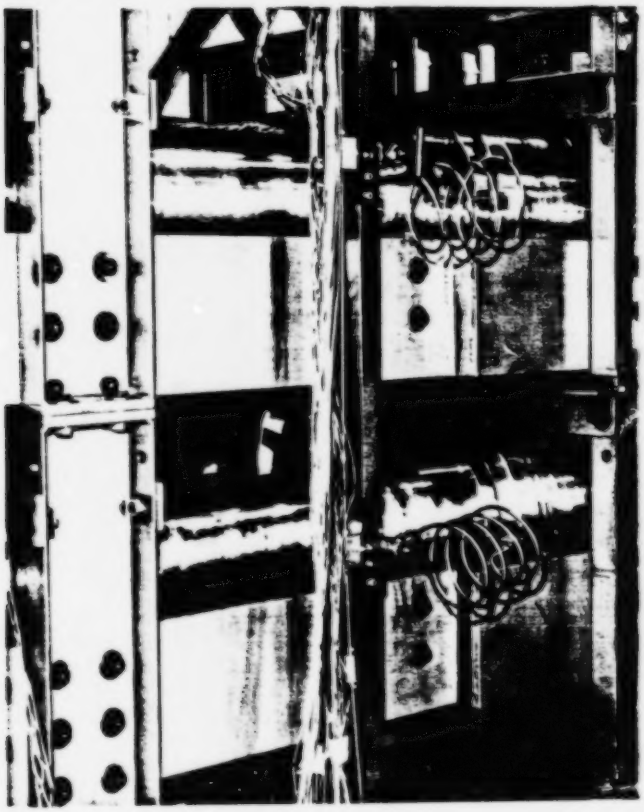


Figure 4. Hydraulic connections to the jack assembly.

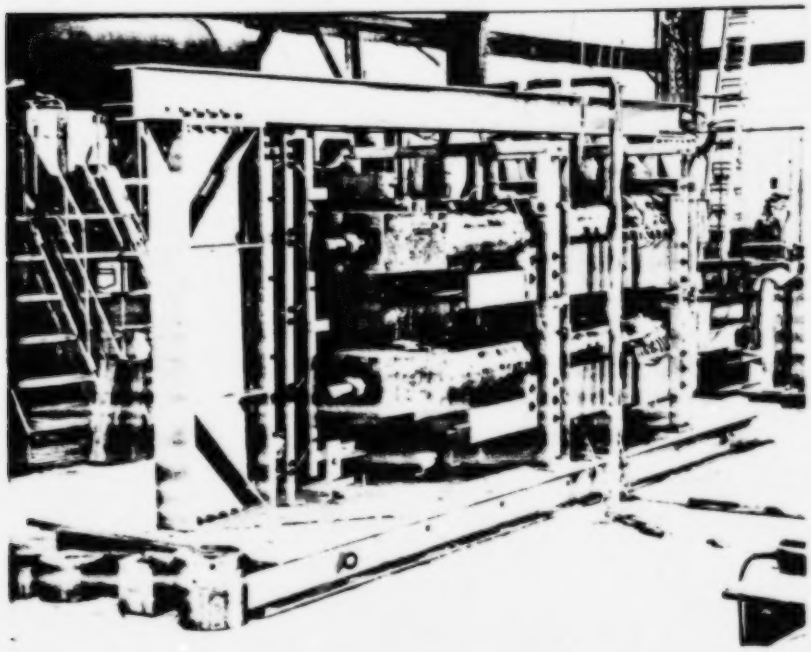


Figure 5. End view showing restraint system and spacer assembly.

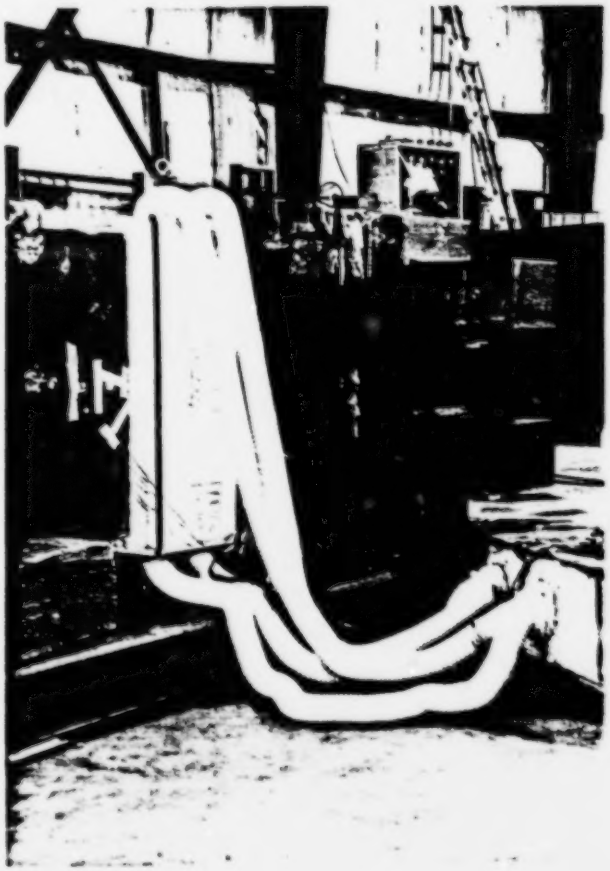


Figure 6. Trials of cooling box using recirculated liquid nitrogen vapour; 75 mm thick plate.

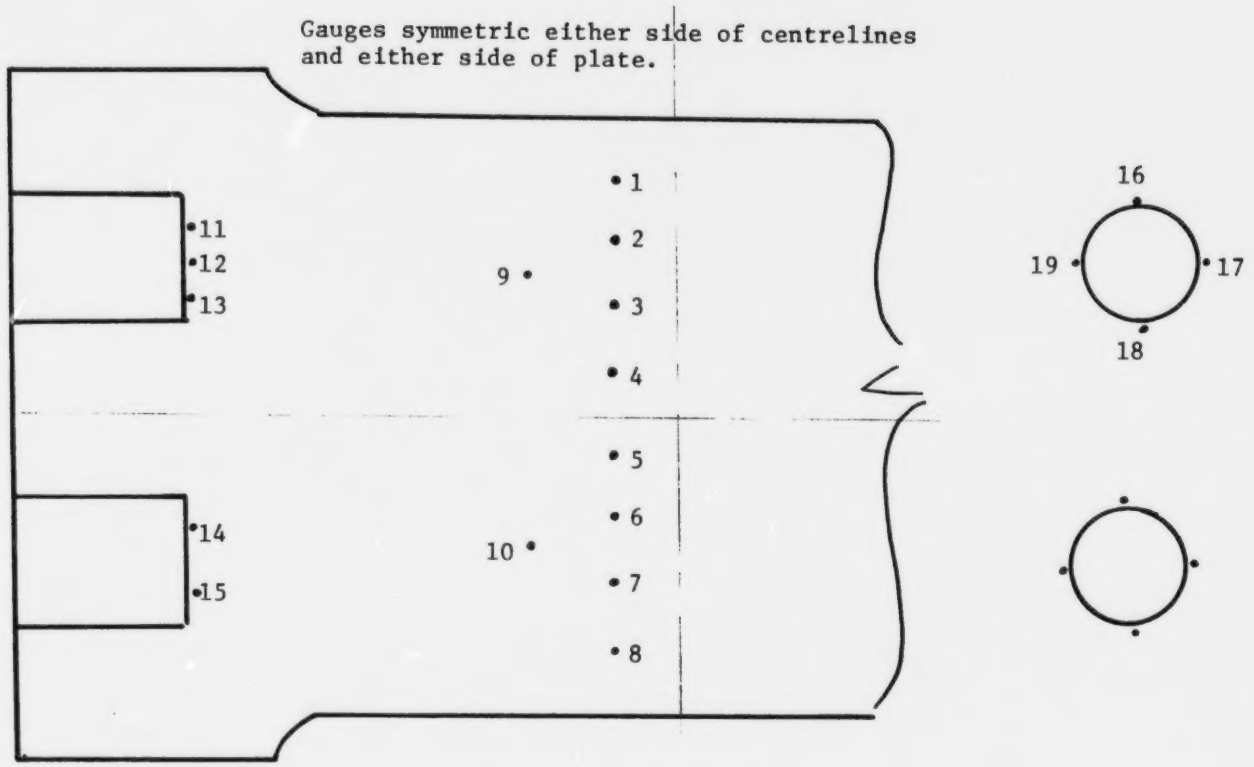
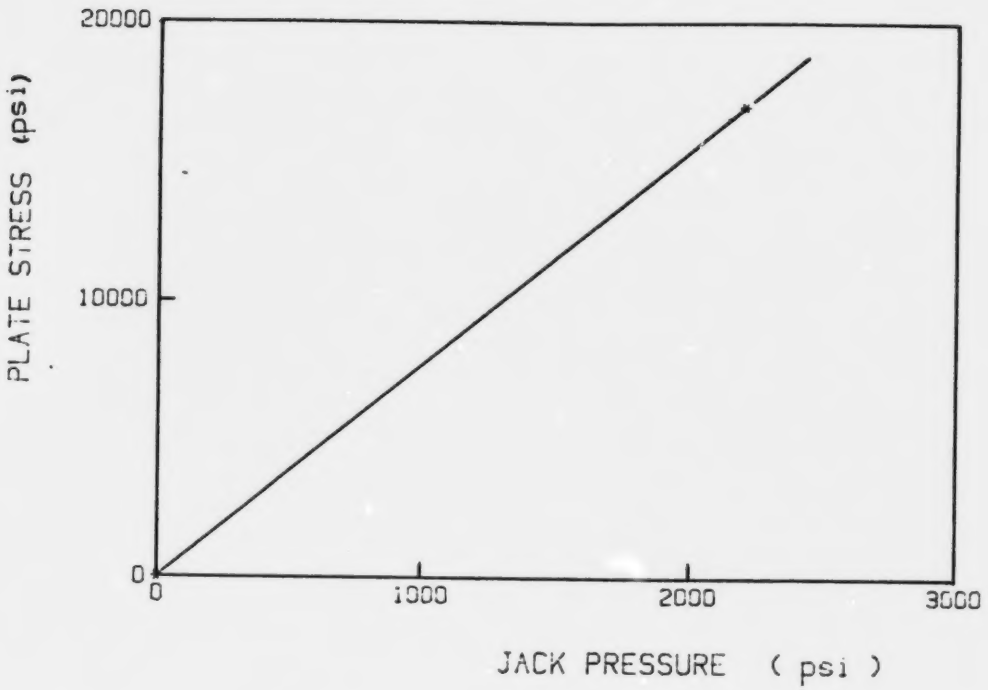


Figure 7. Preliminary instrumentation lay-out on first test.



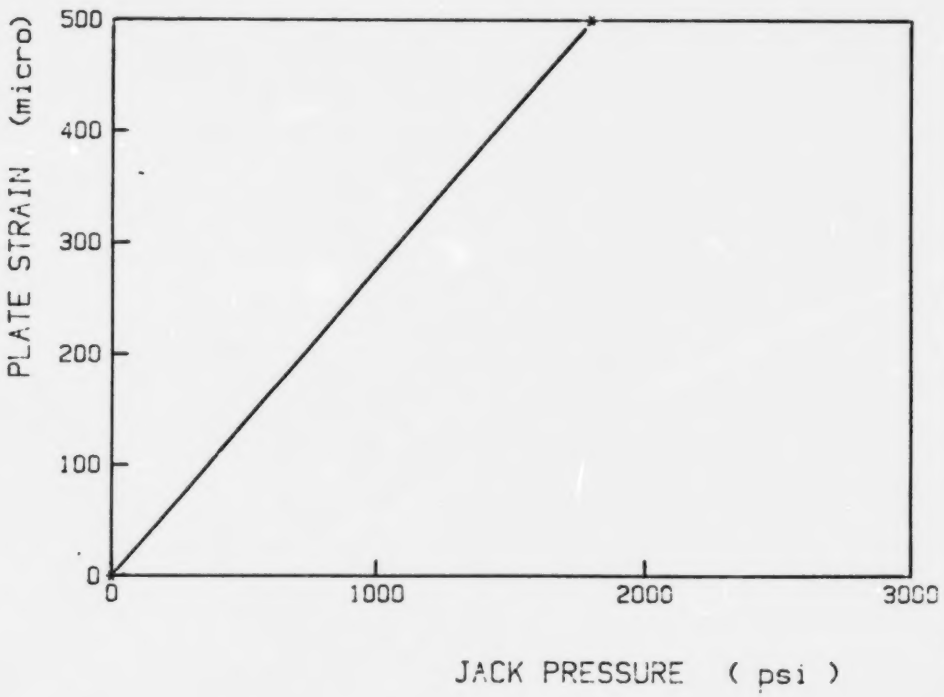


Figure 8. Predicted elastic behaviour of commissioning test plate (25 mm thick).

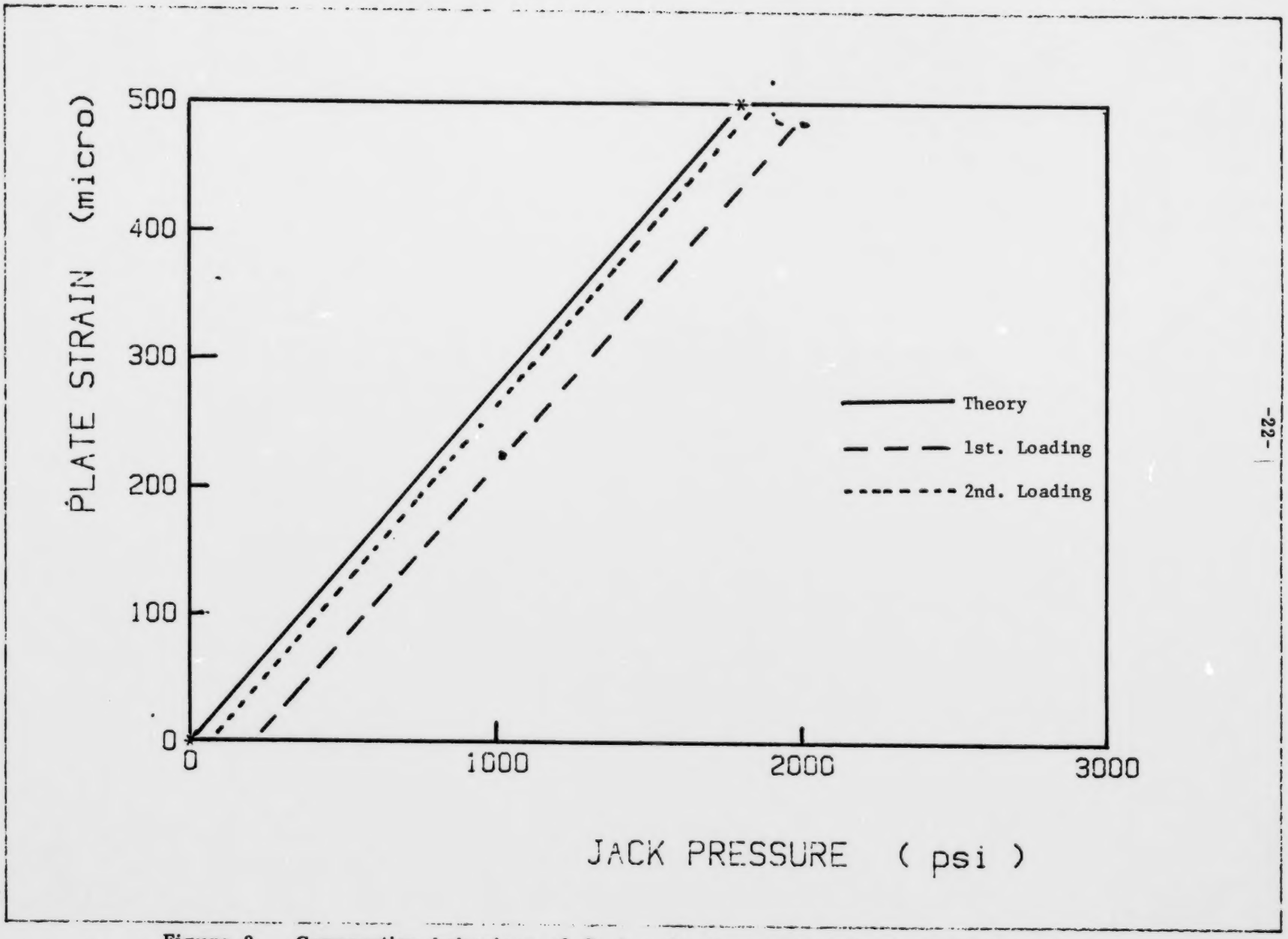


Figure 9. Comparative behaviour of first and subsequent tests in terms of strain versus jack pressure.

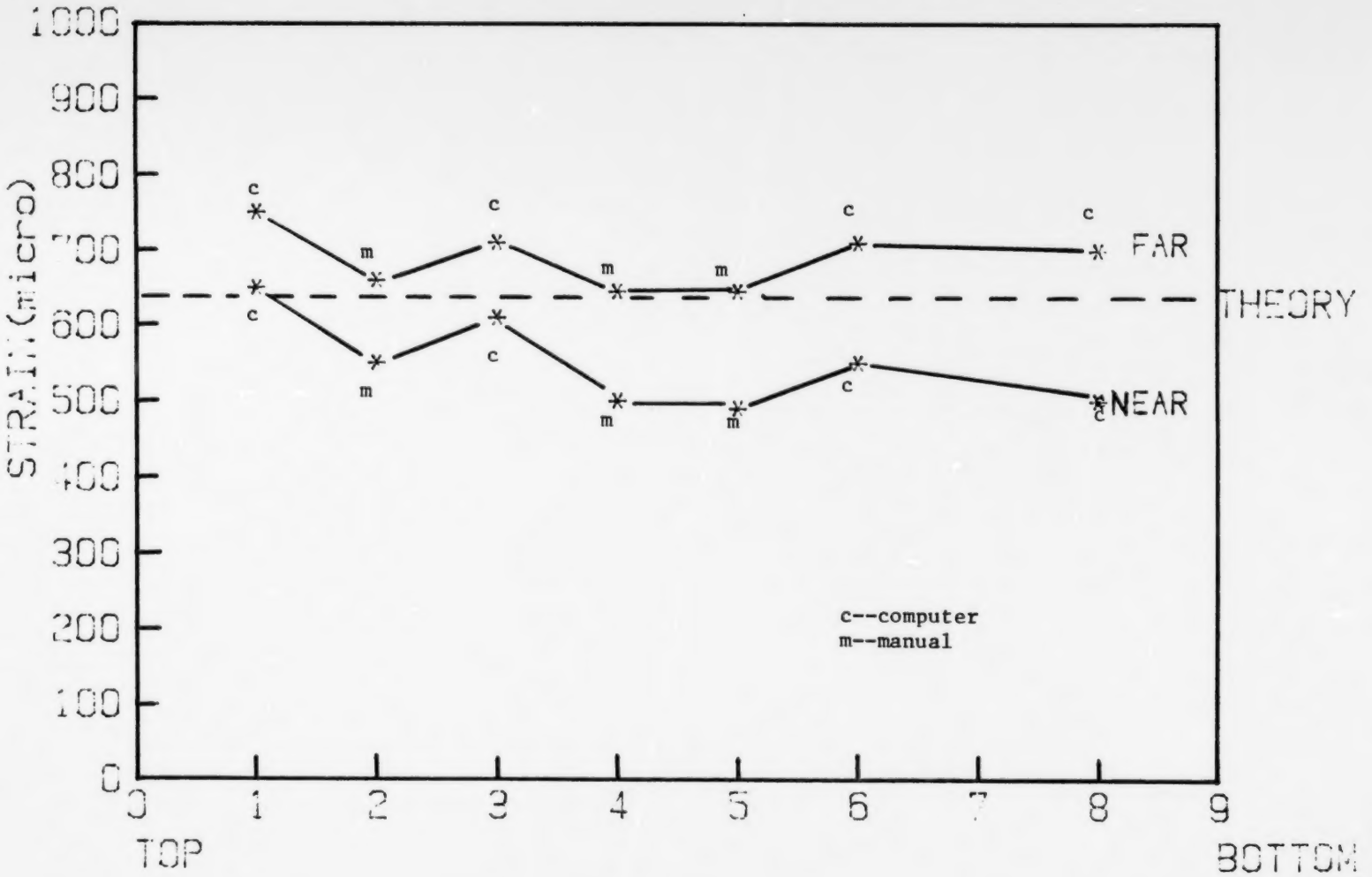


Figure 10. Strain distribution in test plate at 15.2 MPa jack pressure.

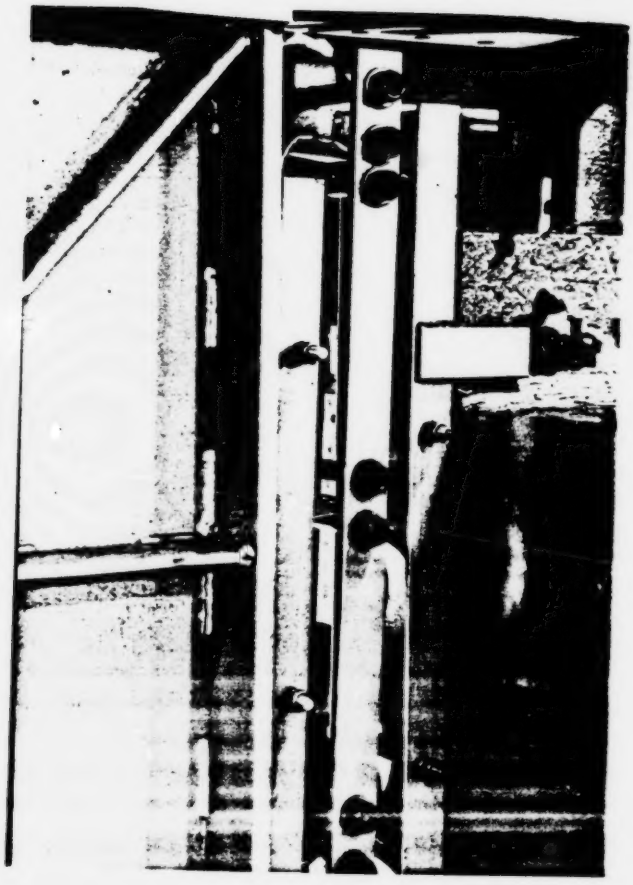
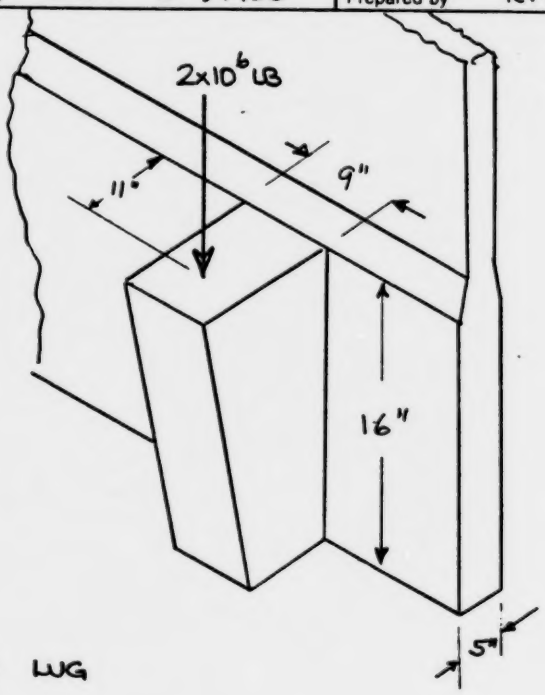


Figure 11. Results of high loading on restraint system.

APPENDIX A

CALCULATION SHEETS

APPROXIMATE STRESSES LUG



CONSIDER END LUG

LUG

AVERAGE SHEAR $= \frac{2\times10^6}{9\times16} = 13888$ PSI

BENDING $= \frac{My}{I} = \frac{2\times10^6(11)(8)}{\frac{1}{12}(9)(16)^3} = 57292$ PSI

MAXIMUM SHEAR $= \frac{3}{2}\frac{V}{A} = 27778$ PSI

PLATE

AVERAGE SHEAR IN WELD $= \frac{2\times10^6\times2}{5\times16\times2} = 25000$ PSI

LOAD FROM BOTH SIDES.

ANALYSIS OF PISTON

END LOAD $= 30000 \frac{\pi}{4} (9.65)^2$

$= 2.2 \times 10^6$ LB

SHEAR STRESS IN OUTER SLEEVE

$= p \frac{a^2}{a^2 - b^2} = \frac{30000 (8.2)^2}{82^2 - 48^2} = 45638$ PSI

THIS STRESS IS TOO HIGH FOR MANY APPLICATIONS. THEREFORE A SHRINK FIT ASSEMBLY OF TWO COMPONENTS WILL BE CONSIDERED. WHILE MORE EXPENSIVE IT SHOULD PROVIDE ADEQUATE FACTORS OF SAFETY WITHIN THE DIMENSIONS DESIRED

WEIGHTS.

a) PISTON ASSEMBLY (4)

$$= .29\left[76\,\frac{\pi}{4}(9.65)^2\right] + .29\left[12\,\pi(8.2^2-4.8^2) + 20\pi(6.8^2-4.8^2)\right]$$

$$= 2518 \text{ LB.}$$

b) PLATE

$$= .29\left[17\times48\times5\times2 + 74\times48\times3\right]$$

$$= 5456 \text{ LB.}$$

c) END LUGS (8)

$$= .29\left[9\times14\times16\right]$$

$$= 584 \text{ LB.}$$

TOTAL PLATE & LUG ASSEMBLY = 10133 LB = 4.6 M TONS.

TOTAL PISTON ASSEMBLY = 10072 LB = 4.6 M TONS

DURATION OF TEST

DEFORMATION OF PISTON RODS

$$= \frac{Pl}{EA} = \frac{2\times10^6\,(76)}{30\times10^6\,\frac{\pi}{4}(9.2)^2} = .076"$$

DEFORMATION OF PLATE

$$= \frac{Pl}{EA} = \frac{8\times10^6\,(76)}{30\times10^6\,(3\times48)} = 0.141"$$

TOTAL PISTON MOVEMENT = .076 + .141 = .217 INCH

ASSUME LUG AND ATTACHMENT FLEXIBILITY PROVIDES A FURTHER .283 INCH MOVEMENT OF THE PISTON

TOTAL PISTON MOVEMENT ASSUMED = 0.50 IN

VOLUME OF FLUID REQUIRED $= .5\,(4)\left(\frac{\pi}{4}9.2^2\right)$

$= 133\ IN^3$.

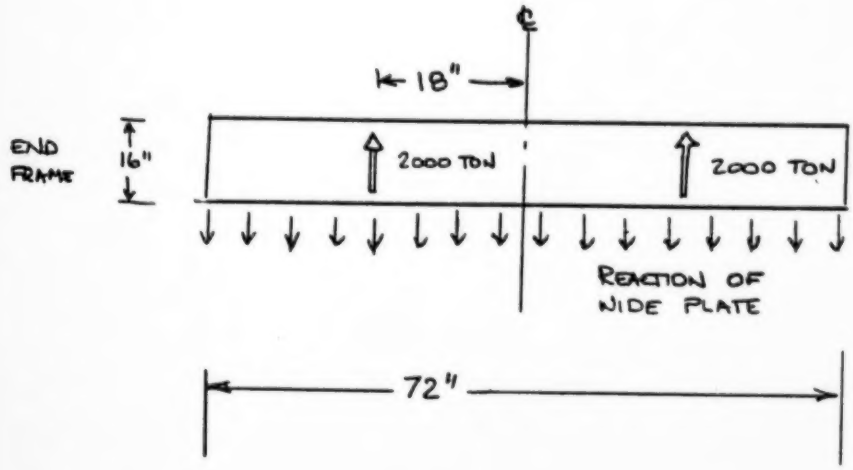
TYPICAL ROUSSEAU PUMP RATING IS 50 IN^3/MIN

THEREFORE TEST DURATION IS ~ 2.7 MINUTES

END FRAME

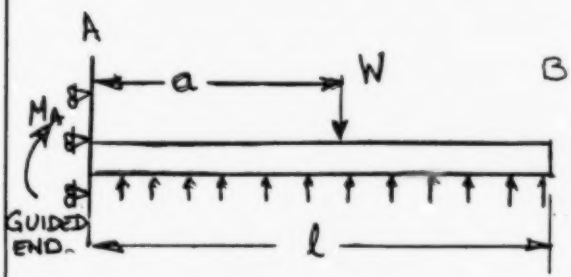
OF SOME CONCERN IS THE STRESS DISTRIBUTION IN THE WIDE PLATE. IT IS DESIRED TO HAVE A CONSTANT UNIAXIAL STRESS. HOWEVER AS THE LOAD IS BEING APPLIED AT DISTINCT POINTS ON THE END FRAMES THEY MUST HAVE SUFFICIENT STIFFNESS TO ENSURE AN EVEN DISTRIBUTION OF LOAD.

CONSIDER THE END FRAME AS A BEAM ON AN ELASTIC FOUNDATION (CONSERVATIVELY NEGLECTING SHEAR EFFECTS AND REINFORCEMENT PROVIDED BY ATTACHMENTS)



AS A CONSERVATIVE CALCULATION ASSUME A WIDTH OF 72 INCHES AND THE JACK LOADS APPLIED 18 INCHES FROM THE CENTRELINE

FROM ROARK, 5TH EDITION, CONSIDER ONE HALF OF THE BEAM



ASSUME

$a = 18''$

$l = 36''$

$\theta_A = 0 \qquad R_A = 0$

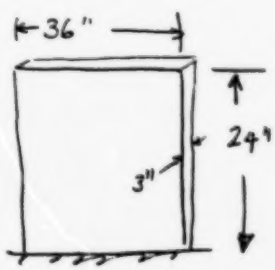
$$y_A = \frac{-W}{4EI\beta^3}\left[\frac{2C_1Ca_1 + C_4Ca_2}{C_{12}}\right]$$

$$M_A = \frac{W}{2\beta}\left[\frac{C_2Ca_2 - 2C_3Ca_1}{C_{12}}\right]$$

$$y = y_A F_1 + \frac{M_A}{2EI\beta^2}F_3 + \frac{R_A^{\,0}}{4EI\beta^3}F_4 - \frac{W}{4EI\beta^3}Fa_4$$

EVALUATING THE CONSTANTS

FOUNDATION MODULUS. $E = 30\times10^6 \frac{LB}{IN^2}$



LOAD = F LB

STRESS = $\frac{F}{3\times36}$

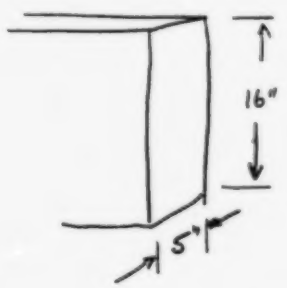
STRAIN = $\frac{F}{3\times36\times30\times10^6}$

DEFLECTION = $\frac{24F}{3\times36\times30\times10^6}$

SPRING CONSTANT $\frac{F}{x} = \frac{3\times36\times30\times10^6}{24} = 135\times10^6 \frac{LB}{IN}$ FOR FULLY WIDTH

$$b_o k_o = \frac{135 \times 10^6}{36} = 3.75 \times 10^6 \text{ LB/IN/IN}$$

CONSIDER THE BEAM



$$I = \frac{1}{12}(5)(16)^3 = 1707 \text{ IN}^4$$

$$\beta = \left(\frac{b_o k_o}{4EI}\right)^{1/4}$$

$$= \left(\frac{3.75 \times 10^6}{4(1707)(30 \times 10^6)}\right)^{1/4}$$

$$= \left(.183 \times 10^{-4}\right)^{1/4}$$

$$= .0654$$

CONSTANTS $\ell = 36$ $\ell - a = 18$ $\beta(\ell - a) = 67.47°$

$$C_{a_1} = \text{Cosh}\,\beta(\ell-a)\cos\beta(\ell-a) = \text{Cosh}(1.177)\cos(1.177)$$
$$= 1.781 \quad (.383)$$
$$= .682$$

$$C_{a_2} = \text{Cosh}\,\beta(\ell-a)\sin\beta(\ell-a) + \text{Sinh}\,\beta(\ell-a)\cos\beta(\ell-a)$$
$$= 1.781 \quad (.924) + 1.473 \quad (.383)$$
$$= 2.209$$

$C_1 = F_1$ at $x = \ell$, $\beta x = 2.35$ $= -3.72$

$C_2 = F_2$ $= .0948$

$C_3 = F_3$ $= 3.685$

$C_4 = F_4$ $= 7.41$

C_{12} at $\beta\ell = 2.35$ $= 27.124$

$W = 4 \times 10^6$ LB $\qquad R_A = 0$

$$y_A = \frac{-4\times10^6}{4(30\times10^6)(1707)(.0654)^3}\left[\frac{2(-3.72)(.682) + 7.91(2.209)}{27.124}\right] = -.02907$$

$$M_A = \frac{4\times10^6}{.0654(2)}\left[\frac{(.0948)(2.209) - 2(3.685)(.682)}{27.124}\right]^{.2094} = -5.431\times10^6$$

$$y = \frac{-5.431\times10^6}{2(30\times10^6)(1707)(.0654)^2} F_3 \quad -.02907 F_1$$

$$\frac{-4\times10^6}{4\,30\times10^6(1707)(.0654)^3} F_{a4}$$

$$y = -.0124 F_3 \; -.698 F_{a4} \; -.02907 F_1$$

$F_3 = \text{Sinh}\,\beta x \sin\beta x$

$F_4 = \text{Cosh}\,\beta x \sin\beta x - \text{Sinh}\,\beta x \cos\beta x$

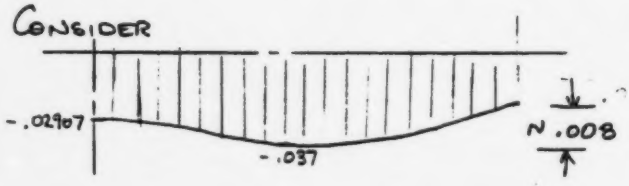
$F_{a4} = \text{Cosh}\,\beta\langle x-a\rangle \sin\beta\langle x-a\rangle - \text{Sinh}\,\beta\langle x-a\rangle \cos\beta\langle x-a\rangle$

$= 0 \qquad x < a$

$\alpha = 18$

x	0	4.58	10.7	15.3	18.3	19.8	24.5	30.6
βx	0	.3	.7	1.0	1.2	1.30	1.6	2.0
F_3	0	.08999	.48869	.98890	1.90688	1.63649	2.37956	3.29789
F_1	1.0	.9986	.96001	.8337	.6561	.5272	−.0752	−1.5656
$\beta(x_1-a)$	−	−	−	−	.0196	.1177	.425	.824
F_{a4}	0	0	0	0	0	.0007	.0524	.375
$-.02907 F_1$	.02907	−.02903	−.02791	−.02424	−.01907	−.01533	+.00219	+.0455
$-.0124 F_3$	0	−.00111	−.00606	−.01226	−.01744	−.0203	−.029445	−.0408
$-.0698 F_{a4}$	0	0	0	0	0	−.0005	−.0036	−.026
Σ DEFLECTION OF END FRAME	−.02907	−.030	−.034	−.036	−.037	−.036	−.0308	−.021

CONSIDER



ASSUME THIS CAN BE ACHIEVED

AVERAGE STRESS = $\frac{4\times10^6}{36\times3}$ = 37,037 PSI

CONSIDER STRESS DUE TO .008 DEFLECTION

$\epsilon = \frac{.008}{36}$

$\sigma = E\times E = 30\times10^6\left(\frac{.008}{36}\right)$ = 6670 psi

THEREFORE THE STRESS IN THE PLATE WILL VARY $\pm \frac{6670}{2}$ PSI = ± 3335 PSI

THEREFORE THE UNIAXIAL STRESS IN THE PLATE WILL BE

37037 ± 3335 PSI

THAT IS THERE WILL BE A ± 10% VARIATION IN STRESS ACROSS THE PLATE

IF THE END FRAME LENGTH (PLATE WIDTH) IS 48 INCHES RATHER THAN 72 INCHES USED IN THE CALCULATION THE RESULTS SHOULD SHOW MUCH LESS VARIATION AND WILL BE MORE ACCEPTABLE.